Thermal-Hydraulic Study for an Engine Cooling Radiator – EVR2021-0031

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Abstract

This work aimed to study the thermal-hydraulic behavior of a small scale radiator in steady state. An experimental bench was assembled with the radiator fit inside a wind tunnel. Air flow in the tunnel was adjusted by a fan and hot water was used as a heat source. It was electrically heated in a tank, pumped to the radiator and then went back to the tank, in a closed circuit. Experiments were carried out with constant water flow whereas air velocity was varied with the fan. The heating power was varied in function of the number of electric heaters turned on inside the water tank. Water and air temperatures were measured with thermocouples. The overall heat transfer coefficient indicated an important influence of frontal air velocity on heat exchange. A dimensionless correlation for air heat transfer coefficient was suggested and the influence of the air velocity at the radiator pressure drop was studied.

Keywords: radiator, heat exchanger, heat transfer, pressure drop

1. Introduction

The field of automotive engine is highly assisted by heat exchangers. The thermal energy from internal combustion that was not converted to work must be rejected to the external environment to avoid engine overheating. Due to this need and to conciliate an efficient heat exchange with vehicle design, reduction of weight and size, and lowest possible cost, compact heat exchangers, known as radiators, are used.

The radiator is a cross flow compact heat exchanger that is composed of flat tubes with louvered fins between them. After being heated by the engine, the fluid is pumped to the interior of the flat tubes, exchanging heat with the air from the environment, that passes through fins, as the vehicle moves. The fins are welded transversely to these tubes to enhance the heat transfer.

It becomes extremely important to know how radiators are affected and what project is necessary to meet the cooling capacity required. Experimental test via wind tunnel is a typical method to study heat exchangers. An open circuit wind tunnel was developed by Wang et al. (1997), to study the heat transfer and friction characteristics of a heat exchanger by varying its geometric parameters. Another experimental investigation is found in Wang et al. (2015) where the air side performance of the heat exchanger was studied. Steady state behavior of automotive radiators placed in the wind tunnel was carried out to evaluate the thermohydraulic performance of the heat exchanger. In Cuevas et al. (2011), the author's experimental results were compared with the classical correlations given in the literature. Karthik et al. (2015) investigated the air side performance of the test radiator and presented the heat transfer and flow characteristics results.

The present work aims to study the thermal-hydraulic behavior in steady state conditions of a radiator inside a wind tunnel when subjected to different air supply flows and heating power to the circulating water. An analysis of the air side is proposed to evaluate the thermal performance of the present heat exchanger and obtain a final non-dimensional correlation. To evaluate the hydraulic behavior of the radiator for the air side, an analysis to the pressure drop is proposed.

2. Methodology

2.1 Experimental procedure

In this work, steady state experimental tests were taken in a typical radiator used in the automotive industry as Formula SAE competitions. It is made of aluminum with fins between flat tubes. It has 35 tubes and 36 rows of fins. The total heat exchange surface area is 1.035 m2 and it has frontal dimensions of (0.146 x 0.178) m.

For the analysis, an experimental bench was assembled in the wind tunnel with the radiator inside as shown in Figure 1. It is composed of 2 circuits, the primary and the secondary. The primary circuit is closed, and it is represented by the water which is pumped with a constant flow from a tank to the radiator, passes through its tubes, and returns to the water tank. The water inside the tank is heated by up to 5 electric heaters of 1 kW each, allowing to control the water temperature at the radiator inlet. The secondary circuit is open and occurs in the wind tunnel, where the atmospheric air flows, passing through the radiator fins. The air velocity is controlled by wind tunnel fan frequency variation.

Figure 1: Experimental apparatus

The experimental bench was outfitted with several measurement instruments for the data acquisition, such as temperature, pressure, and mass flow rate.

The water flow pumped was constant, equal to 4.18 L min^{-1}, measured by the time to fill a certain volume. The air velocity in the wind tunnel varied from 1.27 to 5.93 m s^{-1}. It was measure by a Pitot tube and an inclined liquid manometer (Dwyer, 3 % accuracy).

Thermocouples (type T, Omega, the limit of error of 0.7 °C verified in the test range with a standard reference) were placed at the test bench to measure the inlet and outlet of water and outlet air of the radiator. Due to the non-uniformity of the air after leaving the radiator, a net of 15 thermocouples was used in the wind tunnel (based on the works of Wang et al. (2015) and Glazar et al. (2015)). The inlet air temperature was a constant of 20 ºC, which was obtained by measuring the ambient temperature. The acquisition of the thermocouples was performed by a National Instrument CompactDaq with 6 modules NI 9211.

A total of 15 tests were considered to characterize the radiator. after reaching the steady state condition, which took about 30 minutes among each test, for a certain heating power. After reaching steady state, it was set a time of 3 minutes of data acquisition for each test. The operational conditions to evaluate the thermal behavior of the heat exchanger are given in Table 1.

Table 1: Operational conditions

Air velocity range (m/s)	1,270 - 5,930
Interval scale (m/s)	1,165
Power load range (kW)	3,0 - 5,0
Interval scale (kW)	1,0
Air inlet temperature (°C)	20
Water flow (L/min)	4,18

The authors proposed a simplified analysis for the hydraulic behavior, hence the pressure drop tests were conducted without heating power into water, which means at environment temperature. As the air flow increases through the radiator, their pressure drops are measured using an inclined liquid manometer. A total of 5 tests were carried out.

So the operational conditions to evaluate the pressure drop for the air side through the heat exchanger are the same as Table 1, but without the need of power load.

2.2 Mathematical model

To analyze the radiator behavior in steady state condition, it was set a control volume given in Figure 2.

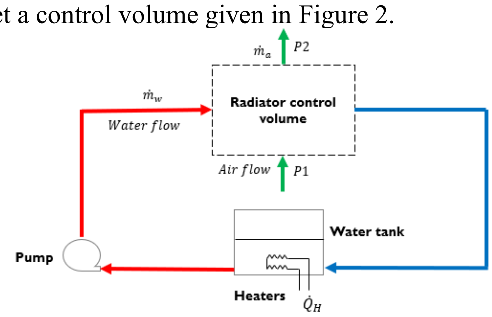

Figure 2: Model control volume

2.2.1 Thermal analysis

Energy balances from the first law of thermodynamic Eq. (1) were applied to the radiator control volume, considering the fluid's kinetic and potential energy negligible. The energy balances allow obtaining the thermal loads of air and water. Their thermodynamics properties were assumed at the mean temperature between the radiator inlet and outlet.

$$\dot{q}_{rad} = \dot{m}_w c_{p,w}(T_{w,i} - T_{w,o}) = \dot{m}_a c_{p,a}(T_{a,o} - T_{a,i}) \qquad (1)$$

where: $\dot{q}_{rad}$ is the heat transfer rate of the radiator [W]; $\dot{m}_w$ is the water mass flow [kg s^{-1}]; $\dot{m}_a$ is the air mass flow [kg s^{-1}]; $c_{p,w}$ and $c_{p,a}$ are the specific heat at constant pressure of water and air respectively [J kg^{-1} K^{-1}]; $T_{w,i}$ is the water temperature at the radiator inlet [°C]; $T_{w,o}$ is the water temperature at the radiator outlet [°C]; $T_{a,i}$ is the air inlet temperature [°C]; $T_{a,o}$ is the air outlet temperature [°C].

The radiator overall heat transfer coefficient is experimentally calculated by the method of logarithmic mean temperature difference in Eq. (2).

$$\dot{q}_{rad} = U_{exp} A \Delta T_{ML} \qquad (2)$$

where: U_{exp} is the experimental radiator overall heat transfer coefficient [W m^{-2} K^{-1}]; A is the radiator total heat exchange surface area on the air side [m^2]; ΔT_{ML} is logarithmic mean temperature difference [ºC].

The non-dimensional correlation involving thermal-flow characteristics of the air side was studied as shown in Eq. (3). Then the overall heat transfer can be estimated, assuming no incrustations and neglecting the thermal conduction as in Eq. (4).

$$Nu = a_1 Re^{a2} Pr^{a3} \qquad (3)$$

$$\frac{1}{U_{cal} A} = \frac{1}{h_i A_i} + \frac{1}{h_o A_o} \qquad (4)$$

where: Nu is Nusselt number; Re is Reynolds number, Pr is Prandtl number and a_1, a_2, a_3 are coefficients; U_{cal} is the estimated radiator overall heat transfer coefficient [W m^{-2} K^{-1}]; A_i is the heat exchange surface area for the water side [m^2]; A_o is the heat exchange surface area for the air side [m^2]; h_i is the heat transfer coefficient of the water side [W m^{-2} K^{-1}]; h_o is the heat transfer coefficient of the air side [W m^{-2} K^{-1}].

For the air side, the value of Nusselt number is based on hydraulic diameter and calculated from Eq. (5) whereas the Reynolds number based on hydraulic diameter as well is obtained from Eq. (6) as the way used in Karthik et al. (2015).

$$N_u = \frac{h_o D_h}{k} \qquad (5)$$

where: D_h is the hydraulic diameter in the air direction [m]; k is the thermal conductivity of the air [W m^{-2} K^{-1}].

$$R_e = \frac{G D_h}{\mu} \qquad (6)$$

where: G represents the total air mass flux [kg m^{-2} s^{-1}]; μ is the dynamic viscosity [N s m^{-2}].

In order to fit estimated overall heat transfer coefficients U_{cal} to experimental values U_{exp}, water heat transfer coefficient h_i and the parameters a_1 and a_2 were evaluated by the least square's method, in Excel Solver. According to literature information, a_3 was considered equal to 1/3 (Gut, 2003) and the initial value of the water side heat transfer coefficient was adopted 1500 W m^{-2} K^{-1} (Kreith et al., 2014).

2.2.2 Pressure drop analysis

The pressure drop for the air side was evaluated and then friction factors could be obtained for the studied geometry of the radiator. Using an inclined liquid manometer which indicates the fluid height difference and applying Eq. (7), pressure drops for the air side were obtained experimentally.

$$P_2 + h\gamma_{fl} = P_1 + h\gamma_{air} \tag{7}$$

where: P_1 and P_2 are the pressures at inlet and outlet of the radiator for the air side [Pa]; γ_{fl} and γ_{air} are the specific weights of the air and the fluid inside the manometer respectively [Kg L^{-1}].

Through the pressure drop data, it is possible to determine the friction factor f, applying Eq. (8).

$$f = \left({}^{\Delta P}/_{2L}\right)\left({}^{D_h}/_{\rho v^2}\right) \tag{8}$$

where: f is the friction factor coefficient for the air side of the radiator; ΔP is the pressure drop [Pa]; L represents flow length in the air direction [m]; D_h is the hydraulic diameter [m]; ρ is the air specific mass [kg m^{-3}] and v is the air velocity inside the wind tunnel [m s^{-1}].

3. Results and discussion

3.1 Thermal results

Figure 3 represents the radiator thermal rejection obtained by Eq. (1), in the function of the air velocity. For each heating power, it may be noted a linear behavior of the thermal rejection as the air velocity increases. This occurs due to the increase of air mass flow and the generation of air turbulences among the radiator fins as well. For the higher heating power, the thermal rejection of the radiator is higher as well.

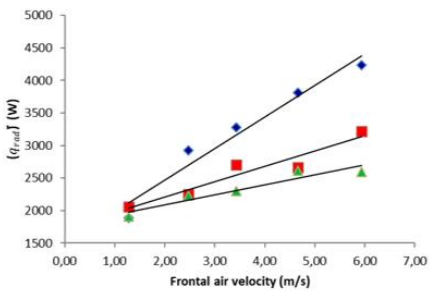

Figure 3: Radiator heat transfer

The heat not exchanged by the radiator was probably lost to the environment through the water hoses and the water tank which has a large surface area.

To find the heat transfer coefficients of the heat exchanger and evaluate them, the overall heat transfer coefficient was obtained by Eq. (2), from experimental data, for each different 15 tests. Thereby the numerical analysis of non-dimensional correlation to the air side from Eq. (3) was able to be carried out.

If the water flow is considered uniform, it can be assumed that its heat transfer coefficient h_i is constant. As the constant h_i and coefficients a_1, a_2 were adjusted, the overall heat transfer coefficient was possible to be estimated by Eq. (4) for each test, in order to obtain the minor square error between the experimental and estimated values. The results are shown in Figure 4.

Figure 4(a) shows the behavior of the heat transfer coefficients of the air side, in the function of its mass flow rate. It is interesting to note that, besides the expected linear increase of the coefficients, their values resemble each other, for each one of the three imposed heating powers. It was obtained a mean correlation of 0.980 among them. It is believed that the small variations are due to environmental adversity and the difficulty to get and control a perfect isolated steady state condition.

The final values obtained by numerical adjustments were h_i = 1721 W m^{-2} K^{-1}, a_1 = 0.146 and a_2 = 0.780. Figure 4(b) shows a comparison between U_{exp} and U_{cal}. There was a good approach between their values, reaching a correlation of 0.982. Thus, the final non-dimensional correlation for the heat exchanger studied is $Nu = 0{,}1459\ Re^{0{,}7801} Pr^{1/3}$

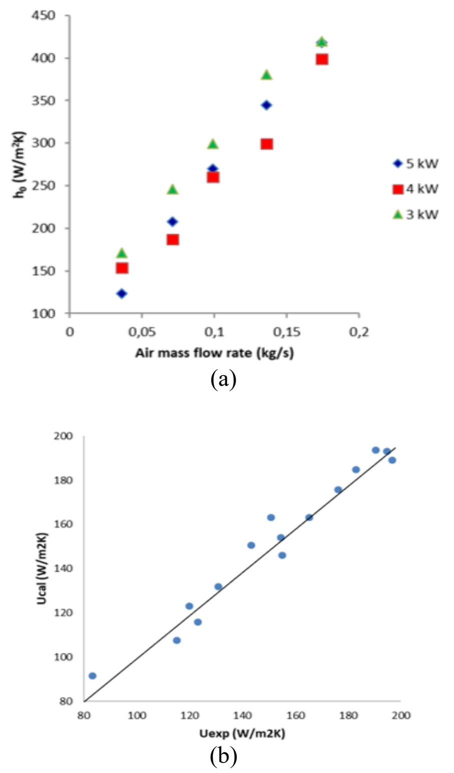

Figura 4: Heat transfer coefficient for the air side (a) and comparison between experimental and calculated overall heat transfer coefficient (b).

3.2 Pressure drop results

As it shows in Figure 5, the pressure drop behavior is given in function of frontal air flow velocity at the inlet of the radiator.

The relationship between the pressure drop and the air velocity was approximated by a second degree polynomial with a correlation of 0,9967. It indicates the pressure drop varies as a function of the square of the air flow velocity as seen in the literature (Kays e London, 1964).

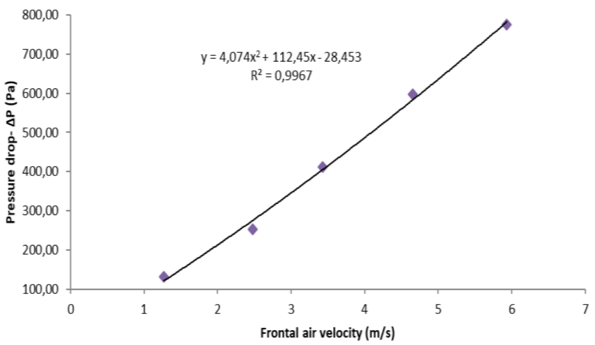

Figure 5: Pressure drop for the air side

Figure 6 shows how the friction factor acts in the function of the Reynolds number for each air flow velocity. It might be seen that exists an inverse proportionality between friction factor and Reynolds number, which means the friction factor is linked to the type of flow on the air side. For low air flow rates and then low Reynold number, the friction factor is high and its value decreases with the increase of the air flow.

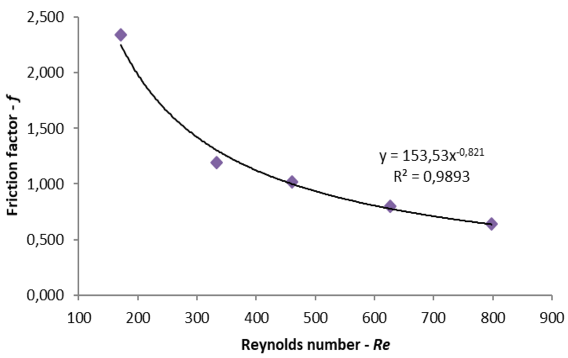

Figure 6: Relationship between friction factor coeficiente and Reynolds number

The increase in pressure drop as a function of air flow velocity is related not to an increase in the friction factor coefficient and energy dissipation by friction, but due to the detachment of the boundary layer and the consequent turbulence generated in the fins. Thus, the generation of vortexes prevents the conversion of kinetic energy from the air into pressure energy, resulting in high pressure drops, the greater the air mass flow.

4. Conclusions

The thermal-hydraulic behavior of a small scale radiator was studied in steady state and the effects of operational parameters have been analyzed. An experimental bench was developed in a wind tunnel divided into two circuits. The experimental results showed that there is a linear relationship between the heat exchange and the radiator air velocity. For the radiator, as a whole, experimental data allows to observe that the overall heat transfer coefficient high values reach nearly 200 W m^{-2} K^{-1}. The linear regression indicated a dimensionless correlation with small deviations from the experimental data. For the pressure drop, the results show the highest values do not exceed 800 Pa, whereas the maximum value for the friction factor is below 2,5. Despite the increase of the radiator heat transfer when the air flow increases, the pressure drop rises as well, and it must be taken into account because it means energy expenditure to pump fluid into the system.

5. Acknowledgments

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6. References

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7. Responsibility notice